JPRS L/10643

8 July 1982

West Europe Report

SCIENCE AND TECHNOLOGY

(FOUO 14/82)

FBIS FOREIGN BROADCAST INFORMATION SERVICE

NOTE

JPRS publications contain information primarily from foreign newspapers, periodicals and books, but also from news agency transmissions and broadcasts. Materials from foreign-language sources are translated; those from English-language sources are transcribed or reprinted, with the original phrasing and other characteristics retained.

Headlines, editorial reports, and material enclosed in brackets [] are supplied by JPRS. Processing indicators such as [Text] or [Excerpt] in the first line of each item, or following the last line of a brief, indicate how the original information was processed. Where no processing indicator is given, the information was summarized or extracted.

Unfamiliar names rendered phonetically or transliterated are enclosed in parentheses. Words or names preceded by a question mark and enclosed in parentheses were not clear in the original but have been supplied as appropriate in context. Other unattributed parenthetical notes within the body of an item originate with the source. Times within items are as given by source.

The contents of this publication in no way represent the policies, views or attitudes of the U.S. Government.

JPRS L/10643

8 July 1982

WEST EUROPE REPORT

SCIENCE AND TECHNOLOGY

(FOUO 14/82)

CONTENTS

 [III - WE - 151 S&T FOUO]

ENERGY

BRITISH SLAGGING GASIFIER METHOD FOR COAL GASIFICATION

Leinfelden-Echterdingen ERDOEL & KOHLE-ERDGAS-PETROCHEMIE in English May 82 pp 219-221

[Text] A 20-year programme to develop a slagging gasifier is now recognized as probably the most-efficient method yet devised to make gas from coal. The slagging gasifier can be used to make substitute natural gas, and to make medium-calorific-value gas for power generation, or to make reducing gas and fuel gas for iron and steel manufacture, as well as systhesis gas for chemical manufacture.

It is known as the British Gas/Lurgi device, and it uses British and German technology and research financed by British Gas, American utilities and EC grants, and is currently ready for use under licence, with full commercial guarantees.

In December 1981, potential customers from more than a dozen countries saw the final stages of the proving run at the Westfield Development Centre in Scotland. At the completion of its run, the new gasifier produced 1750×10^6 ft^3 (49.5×10^6 m^3) of gas from nearly 27 000 tonnes (30 000 US tons) of coal over 90 days.

The gasifier, which has developed from the Lurgi gasifier modified to operate under slagging conditions, was designed to operate at a temperature of 1500 °F (815 °C), above the melting point of ash, thus allowing the ash to be drawn off as molten slag. Because it requires less steam, it gives a higher thermal efficiency and up to four times the gas output of the original Lurgi. An overall coal-to-substitute natural gas efficiency of about 70 per cent is possible. Over the last eight years, the new gasifier has completed a wide-ranging programme proving its commercial viability. During this period, a wide variety of British and American coals have been successfully tested. Many of them had previously been considered difficult, if not impossible, to gasify.

At the start of the system, coal is fed to the gasifier via a lock hoppering device. This is a well established method which has proved troublefree in operation. The system is capable of handling the higher throughputs required by the slagging gasifier.

Fixed-bed gasifiers have traditionally been thought to require sized coal to promote good bed loading and heat exchange in the counter-current flow regime. Thus, the requirements are often quoted as a specific size range with no fines. During trials by the Electric Power Research Institute (EPRI) of the USA, Pittsburgh 8 coal with up to 25 per cent by weight fines of less than 0.6 mm size was fed to the bed top without any significant change in gas analysis. Recent experience at Westfield Station of British Gas has shown that the above figure can be increased by 35 per cent. The slagging gasifier also offers the potential of injecting fine coal directly into the reaction zone of the gasifier via the tuyères, and this technique has also been operated at Westfield. It is expected that this will lead to even greater potential fines intake into the slagging gasifier.

The fixed-bed gasifier generates tar and oils which are carried out from the gasifier with the make gas. In the slagging gasifier they can be recycled to extinction through the tuyères, as was demonstrated at Westfield during the EPRI trials. Tar plays a useful role in aiding the smooth operation of the gasifier by wetting the dust carried over from the top of the bed. The tar laden with dust is removed from the cooling systems and recycled to the top of the fixed bed. As no carbon is taken out by the slag-removal system, the carbon gasification in the slagging gasifier is effectively 100 per cent.

It has been claimed that the fixed-bed gasifiers do not work well with swelling coals. Since 1945 Lurgi has paid much attention to the problem of stirrer design which has much benefited the Westfield Slagging Gasifier. Substantial quantities of strongly caking and swelling coals such as Pittsburgh 8 and Ohio 9, as well as the equivalent strongly caking British coals, have been gasified. No appreciable performance difference has been noted between weakly caking and strongly caking highly volatile bituminous coals.

SNG production

Substitute natural gas can be made at high thermal efficiency from gasifier gas, partly because nearly half of the methane in the SNG is made in the gasifier. The remainder is produced from the carbon monoxide and hydrogen in the gas by reaction with steam over an active nickel catalyst. An upgrading route has been developed, called the high-carbon-monoxide (HCM) process. It has been designed to take advantage of the particular composition of the slagging gasifier gas. In this way the high efficiency of the gasifier is not dissipated in the following stages, and an overall coal-to-SNG efficiency of about 70 per cent is possible. The combination of slagging gasifier and HCM also offers a relatively low capital cost. The process steam for the reaction is generated in a saturator using low-grade heat from cooling trains, and this leads to high thermal efficiency.

Medium BTU Gas

The gas from the slagging gasifier, after removal of tar and aqueous liquor, is suitable for use as a medium BTU fuel gas once it has passed through a desulphurisation stage such as Rectisol. It may not be necessary to remove all the sulphur compounds, and some slip of the more-difficult-to-remove carbonyl sulphide may be economically desirable. Removal of hydrogen sulphide (H_2S) to quite low levels is expected to be mandatory.

Power generation

The direct use of coal for power generation has technical and, particularly, environmental drawbacks, such as the difficult desulphurisation of the coal or the stack gases to reduce atmospheric pollution. It is more attractive to gasify the coal and then to remove sulphur from the crude product gas. However, the removal of sulphur dioxide from stack gases has several disadvantages. By allowing the use of advanced power generation cycles, gaseous fuels can be used more efficiently than the initial feedstock, giving improved thermal efficiency for power generation. Thus, electricity generation, using combined cycles, that is, using an optimised combination of gas and steam turbines to drive alterators, can result in an overall efficiency, including gasification, of above 40 per cent. This compares with less than 35 per cent for conventional steam-cycle power plants fitted with stack gas clean-up devices.

After funding tests at Westfield the slagging gasifier appeared suitable for combined cycle operation. The study had shown that integrated coal-gasification plants using a combined cycle offer distinct environmental advantages and are economically competitive with conventional coal-fired plants.

Another advantage of the gasifier in power generation is its ability to respond quickly to load changes and to run steadily at a variety of loads. The gasifier has been proven to run steadily at all loads used between 30 per cent and 100 per cent of full load and could change rapidly from one load to another within this range. It can be shut down to hot standby in a matter of minutes, retained in this condition for at least 48 hours, and then restarted. Start up, going from an empty reactor to gas making, is easily achieved in four hours, using straightforward procedures.

A mass of experience has been gained about the slagging gasifier from both sponsored programmes and the detailed proving runs of British Gas itself. A confidence exists that Britain has a significant lead ahead of existing and alternative high-pressure coal-gasification systems.

By the 21st century, the main source of energy in many countries is likely to be substitute natural gas made from coal using electricity produced from nuclear power. It is widely acknowledged that production of gas from coal is, at this moment, significantly more efficient than using that same coal to produce electricity.

Research to date has been concentrated on the highly efficient 6 ft (1.82 m) diameter slagging gasifier at Westfield. A larger nominal 8 ft (2.44 m) diameter slagging gasifier is currently being built and will be ready for operation early in 1983. A purpose-built commercial unit of 8 ft (2.44 m) diameter should be capable of gasifying 525 to 725 tonnes (600 to 800 US tons) of coal a day.

Licences for slagging gasifiers up to 8 ft (2.44 m) in diameter are available with full commercial guarantees. British Gas, as sole licensor of the process, will supply the complete design package in association with Lurgi.

CSO: 3102/331

TRANSPORTATION

VOLKSWAGEN WORKING TO DEVELOP GAS TURBINE FOR AUTOMOBILES

Stuttgart MTZ MOTORTECHNISCHE ZEITUNG in German Mar, May 82

[Mar 82 pp 125-128]

[Article by Peter Walzer, Joachim Meier-Grotrian, Rolf Buchheim: "Development Work on an Automobile Gas Turbine"]

[Excerpts] A 100-kW automotive gas turbine is being investigated at Volkswagenwerk AG. The paper describes the special features of the component elements of this engine and presents the current status of engine development in regard to response time, fuel consumption, multifuel capability and engine emissions. Test results of an experimental gas-turbine-powered vehicle are submitted. Reference is made to further improvements in fuel economy with the aid of ceramic components.

1. Development Objectives

As part of the research into alternative power units for the automobile, the gas turbine is one of the units being tested at Volkswagen. Theoretical considerations and simulation calculations show that the gas turbine enjoys advantages with respect to its multifuel capability and absence of exhaust pollutants. Its throttle response, fuel consumption at small part loads and manufacturing costs seem to be problems, but not insoluble ones. This paper describes different steps in development in the attempt to make the gas turbine an alternative power unit that is competitive with the gasoline engine. This objective will have been reached when a successful high-temperature engine is finally developed on the basis described here [1].

2. Engine Construction

Figure 1 shows the construction of the engine. It is a twin-shaft gas turbine with regenerative air preheating and integral reduction gearing. The gas generator and the power turbine work in the longitudinal axis of the engine. The gas generator drives the compressor and, through a secondary takeoff, the various auxiliaries needed to operate the engine. The power turbine drives the output shaft through reduction gearing. The tubular combustion chamber sits on top of the engine. Two regenerator discs are located on the sides as heat exchangers. The engine has adjustable guide vanes ahead of the compressor and the power turbine. It is equipped with an electronic-hydraulic control system.

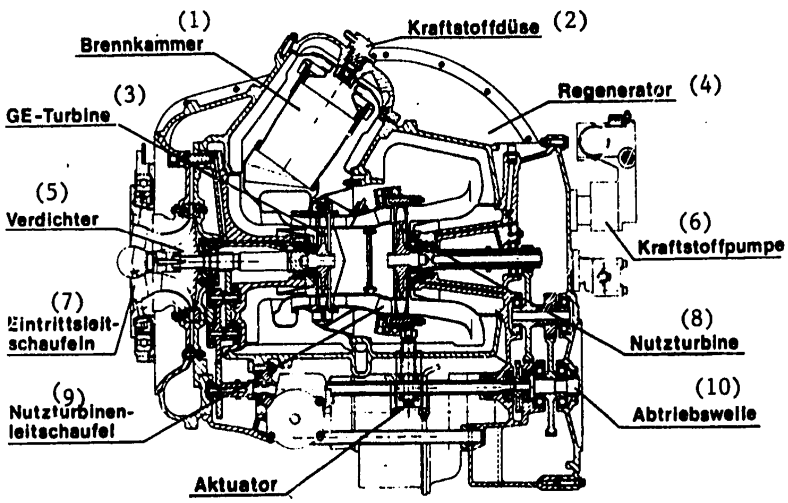

Bild 1: VW-Gasturbine GT 150 für Personenwagenantrieb

Fig. 1: VW-GT 150 baseline gas turbine for cars

Figure 1. VW Gas Turbine for Passenger Cars

Key:

1. Combustion chamber
2. Fuel nozzle
3. GE turbine
4. Regenerator
5. Compressor
6. Fuel pump
7. Inlet guide vanes
8. Power turbine
9. Power turbine guide vanes
10. Output shaft

Table 1 contains additional important data on engine layout. The engine's rated power is 100 kW, working with a pressure ratio of 4.5:1 and a turbine intake temperature of 1,283 K. The maximum speed of the gas generator is 63,700 rpm and that of the power turbine is 52,200 rpm. The output shaft reaches a speed of 5,500 rpm, which is normal for piston engines. Engine volume and weight are comparable to those of a gasoline engine of equal power. Development of the prototype of this engine was carried out between 1972 and 1975 in conjunction with the Williams Research Corporation, in Walled Lake in the United States. The rest of the paper describes mostly the work carried out as part of the research project sponsored by the BMFT [Ministry for Research and Technology] from 1977 to 1980 and the improvements that were achieved.

2.1 Compressor with Backward Curved Rotor Blades and Adjustable Intake Ring

In order to obtain an overall high thermal efficiency with the engine, a compressor is needed which has such a broad operating range that even at partial load the vehicle can be driven with the engine operating at the highest possible temperatures,

Table 1. Full-load specifications of the GT 150 vehicle gas turbine

Air flow	0.84 kg/sec
Pressure ratio	4.5:1
Turbine intake temperature	1,283 K (1,010°C)
Gas generator speed	63,700 rpm
Power turbine speed	52,200 rpm
Output speed	5,500 rpm
Intake guide vane adjustment	Power turbine
Control system	Electronic
Engine volume	0.275 m^3
Weight	210 kg

that is, working temperatures which can just be tolerated by the materials of the turbines and heat exchangers. In addition, optimal efficiency should, first of all, be in the low partial load range and, secondly, always be close to the operating curve. Radial compressors with backward curved blades meet these requirements better than those with radially curved tips. But in order to obtain the same pressure ratio, higher circumferential speeds, that is, greater rigidity is required. In this case, by choosing a high-strength aluminum alloy and as a result of careful three-dimensional finite-element analysis, a compressor with backward curved blades was successfully developed, which reaches the desired pressure ratio of 4.5:1 at a circumferential speed of 513 ms/sec.

With an adjustable guide vane ring ahead of the compressor, the airflow in front of the turbine can be directed in such a way that

--the engine's fuel consumption at idle is reduced, while gas generator speed is unchanged,
--specific fuel consumption at part load is reduced,
--during nonstationary operation, changes can be achieved in power output over a particular range simply by rotating these vanes while maintaining constant gas generator speed.

It can be seen from the compressor performance map that a broader operating range was achieved (up to 12 percent at high rpm) and that good efficiency is still available down to low rpm. With a guide vane adjustment equal to 60° constant twist, the same effect is obtained with respect to air and fuel flow in the lower rpm range as with a gas generator speed reduced by 10 percent without any pretwist. For the sake of comparison, the performance map of an earlier compressor with radially curved blades is included.

Figure 5 shows the influence of the adjustable inlet guide vanes on compressor efficiency and airflow. At part load, with 63-percent gas generator speed and a turbine intake temperature (TIT) of 1,120 K, there is an improvement in efficiency of more than 2 percent. At idle speed, with open turbine guide vanes (NGV) [acronym not positively identified], the result is a 30-percent reduction in airflow as a result of reduced pressure ratio, with a vane angle in the inlet guide mechanism equal to 60° pretwist.

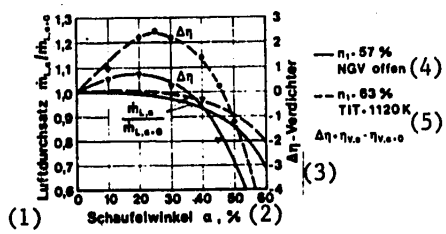

Fig. 5 Effect of adjustable inlet guide vanes

Key:

1. Airflow $m_{L,a} / m_{L,a=0}$
2. Vane angle a, %
3. $\Delta\eta$ Compressor
4. ___ n_1 = 57% inlet guide vanes open
5. _ _ n_1 = 63 % turbine inlet temperature = 1,120 K

2.2 Tubines with Ceramic Shroud

During hot operation, the radial clearance has a great influence on the efficiency of the turbine stages. In a cold engine the clearance between the outside diameter of the rotor and the internal diameter of the shroud is very small. However, after starting, these two components heat up at different rates and to different degrees. Since heat is conducted away through the bearings, the rotor expands less than the turbine shroud, and a large radial clearance is created during engine operation.

When the engine is shut down, the housing with the shroud cools more quickly, while heat is retained longer in the rotor. To prevent any contact under all conditions, it would be necessary to accept an operating clearance of 0.75 mm in a purely metal design, which, with 15 mm-long blades, is the equivalent of a relative clearance of 5 percent. Component efficiency of the turbines can deteriorate up to 10 percent, compared with very much smaller clearances. Sinces measures such as cooling the housing components would entail removing large amounts of heat and such measures would be extremely expensive from a design standpoint, the operating clearance in this engine is maintained at 0.3 mm by using a ceramic turbine shroud with a low thermal rate of expansion.

Figure 6 shows the parts of the ceramic turbine shroud. Reaction sintered silicon nitride (isostatically pressed) was used as the ceramic material. Figure 7 shows how much the efficiency of the gas generator and the power turbine could be increased by introducing these ceramic shrouds.

2.3 Heat Exchanger Matrix and Flow Distributor on the Heat Exchanger

At small part loads and at idle, up to three-quarters of the energy needed to drive the engine is taken from the heat exchanger. Because of the small engine dimensions needed, two regenerator discs mounted on the side were selected. By careful development of the seals, leakage could be reduced to between 2 to 3 percent of the

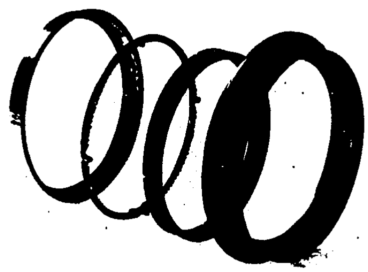

Fig. 6 Ceramic turbine shroud with retainer

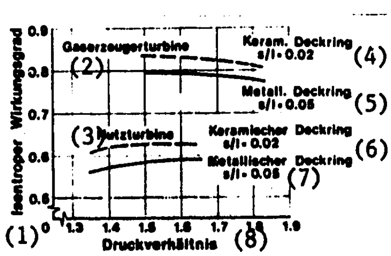

Fig. 7 Turbine efficiency with ceramic shrouds

Key:

1. Isentropic efficiency
2. Gas generator turbine
3. Power turbine
4. Ceramic shroud s/l = 0.02
5. Metal shroud s/l = 0.05
6. Ceramic shroud s/l = 0.02
7. Metal shroud s/l = 0.05
8. Pressure ratio

airflow at a given time, including the leakage carried in the matrix itself. The matrix and the airflow to the matrix were carefully optimized to achieve the highest possible degree of heat exchange.

The regenerators were obtained from the Corning Glass Works and are made of lithium-aluminum-silicate. Table 2 shows clearly the progress that was made by switching from the triangular matrix T 20-38 to the triangular T 14-20 matrix, with respect to the size of the exchanger surface per unit of construction volume. The rectangular matrix listed in the table would be even better, but the manufacturing techniques have not yet been mastered.

Matrix	T 20-38	T 14-20	5.3
Porosität	0.640	0.745	0.786
Zellen/cm^2	148	219	62
Wandstärke mm	0.120	0.070	0.102
Hydr. Durchmesser mm	0.569	0.543	0.762
Spez. Oberfläche m^2/m^3	4500	5490	3707
Triebwerksversuch	x	x	1980

Table 2. Data of Regenerator Matrices

Matrix		T 20-38	T 14-20	5.3
Porosity		0.640	0.745	0.786
Cells/cm^2		148	219	62
Wall thickness	mm	0.120	0.070	0.102
Hydr. diameter	mm	0.569	0.543	0.762
Spec. surface	m^2/m^3	4,500	5,490	3,707
Engine test		x	x	1980

In the engine unequal flow through the regenerator discs reduces the exchange rate and causes increased loss of pressure. Improving the flow distribution is, therefore, an effective measure to increase the efficiency factor of the heat exchanger.

Figure 9 shows the effects of these steps on the efficiency factor of the heat exchanger in the ranges of particular interest here, idle and low part load. It can be seen that it was possible to improve the efficiency of heat exchange by up to 4 points by improving flow distribution and by switching to the T 14-20 matrix.

2.4 Combustion System with Low Pollution Level and Multifuel Capability

The CO-NO_x correlations shown in fig. 10 were measured with diffusion-flame combustion chambers and premix combustion designs. HC emissions are not shown, since they normally do not exceed the limit, if CO emissions are not in excess of the limit. The acceptable operating ranges for 1.5 grams/mile NO_x and 0.4 grams/mile NO_x are shown as dark bands. A gas turbine engine with a fuel consumption of less than 15.7 liters/100 kms can be operated withing the ranges shown.

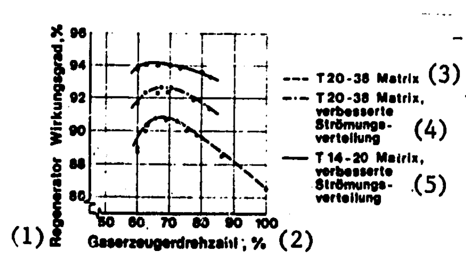

Figure 9. Improvement in Regenerator Efficiency

Key:

1. Regenerator efficiency, %
2. Gas generator speed, %
3. - - - T 20-38 matrix
4. - - T 20-38 matrix, improved airflow distribution
5. ___ T 14-20 matrix, improved airflow distribution

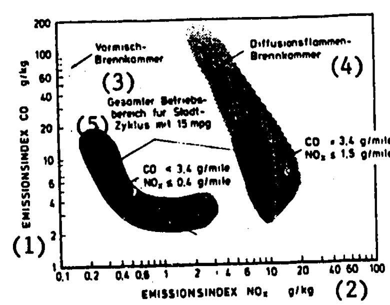

Figure 10. Ranges for Different Emissions Improvement Systems

Key:

1. Emissions index CO grams/kg
2. Emissions index NO_x grams/kg
3. Premix combustion chamber
4. Diffusion-flame combustion chamber
5. Total operating range for city cycle with consumption of 15 mpg

Different combustion systems were developed for the engine in accordance with these emissions levels and used as the emissions limits required, see fig. 11.

(1) (2) (3)

Figure 11. Combustion Chambers for Passenger Car Gas Turbines

Key:
1. Direct injection
2. Single-stage premixing
3. Dual-stage premixing

--On the left is a conventional combustion chamber, optimized for minimum emissions, with direct injection of the fuel into the reaction chamber. With this type of combustion chamber, all the requirements now in force and planned in Europe can be complied with. The U.S. emissions regulations currently in force cannot be met with this combustion chamber. This combustion chamber is extremely cost-effective to manufacture and presents no operating problems in a gas-turbine passenger car.

--In the center is a combustion chamber in which all the fuel is premixed and prevaporized before it goes to the reaction chamber. A continuously burning auxiliary flame is required for operation. All the emissions regulations in effect in Europe and the United States can be met with this combustion chamber. It cannot meet the extremely rigid limits planned in the United States of 0.4 grams/mile for NO_x, 3.4 grams/mile for CO and 0.41 grams/mile for HC (Muskie standards). This combustion chamber can be made only at a considerably higher cost than the combustion chamber shown on the left.

--On the right is a combustion chamber with two-stage injection and premixing. The first stage controls idle and the lightest loads, while the second stage comes in only during acceleration and at higher loads. In both reaction zones the fuel-air mixture is burning close to the lean flame limit. Only with this one of the three combustion systems shown is it possible to meet the Muskie standards, but it requires considerably higher construction costs and a very complex control system.

The combustion process in a gas turbine is, for a number of reasons, particularly suitable for operation on different fuels.

1. The continuous combustion process, which takes place at almost constant pressure, makes it possible to use fuels with a broad range of octane numbers, vaporization pressures, distillation processes, ignition qualities and heat values.

2. The large amount of excess air allows fuels with widely divergent stoichiometric ratios to be used.

3. The reaction volume has free limits and adjusts automatically to the magnitude required for the various fuels.

For these reasons, the combustion chambers developed so far could be operated with a broad range of fuels. More details are given in [2].

2.5 Electronic Control System

The electronic control system of an automobile gas turbine must be able to function satisfactorily under different operating conditions:

1. The load cycle of an automobile gas turbine is characterized by very frequent load changes and frequent driving in the low part load range.

2. The engine must be able to respond to requirements following a load change.

3. Low consumption and low emissions can only be achieved if the various engine components are optimally harmonized with each other.

4. Even an unskilled driver must find it possible to operate the engine safely.

Optimal fuel consumption, the lowest emissions and instant throttle response can be achieved in a gas turbine, if the engine can be operated at the highest possible temperature for the material. Operating in a range which is so close to the limits of permissible material stress requires an extremely precise control system to guarantee maximum operating safety. If there is any danger of these limits being exceeded, the controller must automatically initiate rapid countermeasures.

In cooperation with Lucas C.A.V., London, a control system was developed which meets these different requirements. It consists of the electronic control unit, several sensors for rpm and temperature and a hydromechanical system for the fuel supply and the two mechanisms for adjusting the guide vanes.

Details of the switching circuit and overall function are given in [3]. The following section describes some stationary and nonstationary operations.

In stationary operation, the main control circuit governs gas generator speed according to accelerator pedal position; this is carried out with the help of the "Slave Datum" control loop, patented by Lucas, which influences fuel flow.

The other value controlled in stationary operation is the temperature between the turbine stages. This is achieved by the integral control unit adjusting for zero deviation between the specified temperature which is measured on the one hand and created as a function of corrected gas generator speed on the other; to do this, the control unit adjusts the guide vanes accordingly.

The compressor intake guide vanes are rotated each time to the positions in which the lowest possible fuel consumption at idle on the one hand, and the lowest possible specific fuel consumption under load requirements on the other hand, can be obtained.

The two-stage premix combustion chamber, which is needed for the lowest emissions, is regulated by the control unit so that the first or second fuel nozzle is supplied with fuel according to the fuel-air ratio.

During nonstationary operation, the inter-stage temperature control adjustor is occassionally switched off to obtain optimal response from power turbine torque. This is done by again working, a very short time after a load change requirement, toward engine conditions which are close to the condition that is then to be set.

After initiation of full-load acceleration, gas generator speed is increased in accordance with the "Slave Datum" control circuit; in doing so, maximum permissible acceleration, which is predetermined as a function of gas generator speed, remains limited. This creates temperatures that would not be permissible for stationary operation but which can be tolerated by the engine material for the short duration of acceleration.

By controlling the power turbine guide vanes in nonstationary operation, a high rate of acceleration of the gas generator is achieved on the one hand, and on the other, a rapid increase in torque at the power turbine is obtained during this acceleration.

If the driver removes his foot completely from the accelerator pedal under any load condition, the power turbine guide vanes are rotated rapidly to a braking position, creating negative torque at the power turbine. The braking position is fixed at a value at which the turbine intake temperature is at a safe, low level (the temperature control circuit is open). Not until the power turbine speed has dropped to a fixed value are the power turbine guide vanes returned to the open idle position.

If the driver only partially removes his foot from the accelerator pedal under any load condition, the power turbine guide vanes are rotated to a fixed part-load position, while the gas generator shaft is slowed down, either in accordance with the "Slave Datum" control circuit or in accordance with the deceleration fuel volume, which is to prevent combustion chamber flameout.

Depending on the position of the ignition key, either "Start" or "Off," a start or shutoff sequence runs in the control unit to start or shut off the engine automatically.

The safety logic of the control unit has two tasks: to guarantee maximum protection for the experimental engine and--in the event that some components fail--to make maximum controllability and potential performance possible under safe operating conditions.

The result of these considerations is a "Fail-Safe-System," which allows the car to be driven home under reduced power, if some components fail, and if certain other components fail, the engine is shut down if safe engine operation cannot be guaranteed.

If the turbine shafts are run at excessive speed, and if the guide vane controls (including the position sender) fail, the engine is switched off. If the speed pickups, temperature sensors and accelerator position sender fail, speed and fuel flow are reduced to a low and safe level.

[May 82 pp 185-189]

[Excerpts]

3. Test Stand Results

Figure 13 shows the experimental VW-GT 150 gas turbine on the test stand. After all the components had been developed on special test rigs, the interaction of all the components in the engine was optimized on this engine test stand with a view to the objectives mentioned at the beginning of the paper.

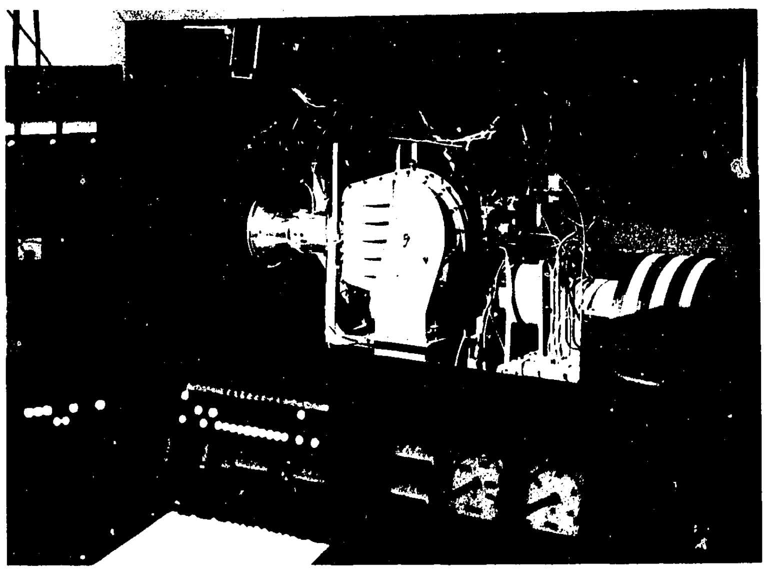

Figure 13. Experimental automobile gas turbine VW-GT 150

3.1 Throttle Response

One of the goals of the development was to be satisfactory engine throttle response. The development work for this had to be the first in time carried out, to ensure that the other steps that are planned to reduce consumption and emissions can build on an engine and a control system that operate satisfactorily. The throttle response of a two-shaft gas turbine is determined mainly by the time required by the gas generator to accelerate from idle to full speed. This time can be influenced through the polar moment of inertia of the gas generator, the rpm difference of the gas generator that has to be overcome or through the excess power at the gas generator shaft.

During the selection of the materials and the aerodynamic design of the gas generator efforts were made to achieve the lowest possible polar moment of inertia, to ensure the best throttle response possible.

Since the lowest idle speed possible is needed, from the point of view of fuel consumption, tests can still be carried out to shorten response time. primarily by increasing surplus power. This surplus can be achieved as a temporary increase in turbine intake temperature and through a temporary increase in the pressure gradient at the gas generator turbine.

The change in gas temperature can be carried out in a few milliseconds by altering the fuel delivery. Upper limits for the excess temperature are set by the pumping limit of the compressor, by the broad flameout limit of the combustion chamber and, if the process lasts for any time, by the temperature resistance of the materials, for example, of the heat exchanger.

An increase in the pressure gradient can be achieved in one-tenth of a second by opening the power turbine guide vanes. The limits to be observed are that, with increasing vane opening, the runup time of the gas generator becomes shorter, while, on the other hand, the torque that is developed at the power turbine during the runup of the gas generator decreases. Finally, it is true of both temperature increase and pressure gradient increase that too large and too abrupt changes can lead to instability in the running of the engine.

Work on improving throttle response is accordingly concentrating on refining fuel delivery and optimizing power turbine guide vane control. In fig. 14, at the top, power turbine guide vane position and fuel flow are shown as functions of time. For comparison, the temperature reached momentarily by a component behind the gas generator and gas generator speed are shown. The graph in each case covers the first 2.5 seconds after initiation of full-throttle acceleration from idle. As fig. 14 shows, up to 150 percent of the amount of full-load fuel is injected in about 1 second, and this amount of fuel for acceleration is being injected very early, while airflow volume is still very low. Simultaneously, the guide vanes are moved from their open idle position to a part-load position, in which a good compromise is reached between pressur gradient at the gas generator turbine and development of torque at the power turbine. Shortly before maximum gas generator speed is reached, the vanes begin to rotate to their final position, corresponding to stationary conditions. The gas temperature that is generated briefly is very high and can be used for acceleration, but in spite of that--as the figure shows--the momentary component temperature does not exceed the permissible limits. Fuel delivery and rotation of the vanes are started in the control system after receiving signals for speed and temperature. Technically, the impulses for rpm can be registered quickly enough, but on account of large time constants in the thermal elements, the gradient of the signal has to be processed.

In fig. 14, at the bottom, the increase in torque is shown as a function of time following the start of acceleration. It can be seen that the goal of 60 percent of rated torque after 0.9 seconds, with satisfactory driving characteristics, was almost reached.

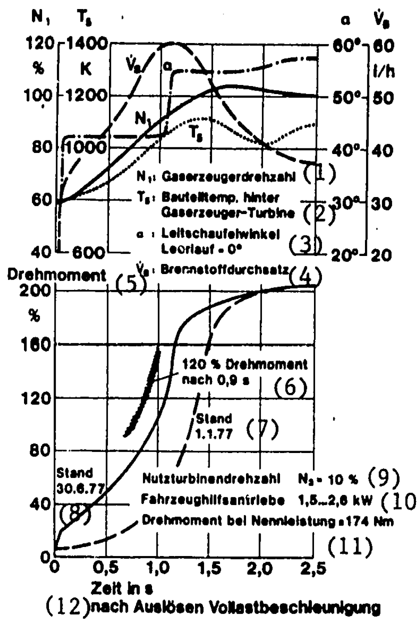

Figure 14. Torque response

Key:

1. Gas generator speed
2. Component temperature behind the gas generator turbine
3. Guide vane angle. Idle = 0°
4. Fuel flow
5. Torque
6. 120% torque after 0.9 secs
7. Status 1 January 1977
8. Status 30 June 1977
9. Power turbine speed N_3 = 10%
10. Vehicle auxiliary drives 1.5-2.6 kW
11. Torque at rated power = 174 Nm
12. Time in secs after initiation of full-load acceleration

3.2 Specific Fuel Consumption

Figure 15 shows the specific fuel consumption for part-load driving on the road as a function of the power required to drive the car. The control system ensures that the guide vanes for the power turbines are rotated in such a way that the highest permissible gas temperature is maintained at the turbine intake under all part-load conditions and that, in each instance, the guide vanes ahead of the compressor create optimal airflow conditions. Figure 15 shows how the specific fuel consumption improved during development by applying the modifications described above to the components. For comparison, the corresponding consumption curve of a carefully optimized gasoline engine is shown. The graph shows that it has been possible to equal or to beat the specific consumption of this engine, down to about 30 percent power output.

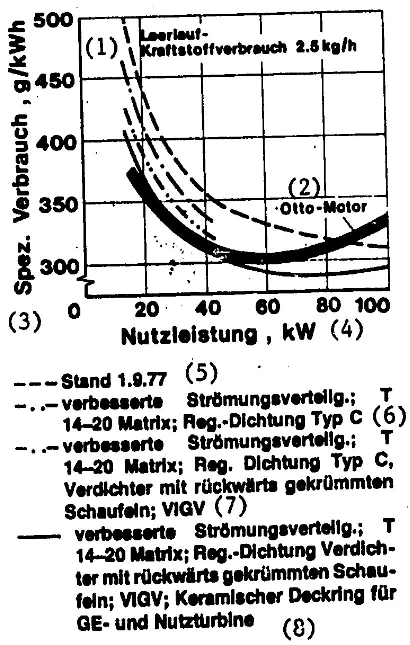

Figure 15. Fuel consumption of the VW-GT 150 vehicle gas turbine, over-the-road, part-throttle operation.

Key:

1. Fuel consumption at idle, 2.5 kg/hr
2. Gasoline engine
3. Specific consumption, grams/kW hour
4. Rated power kW
5. - - - Status 1 September 1977
6. -.- Improved flow distribution; T 14-20 matrix; regulator seal Type C
7. -..- Improved flow distribution; T 14-20 matrix; regulator seal Type C, compressor with backward curved blades; VIGV [variable inlet guide vanes]
8. —— Improved flow distribution; T 14-20 matrix; regulator seal [Type C], compressor with backward curved blades; VIGV; ceramic shroud for the GE and power turbines

Consumption at idle is very important for a passenger car in a city driving cycle. In regard to low consumption, the idle speed chosen should be very low, in regard to a short response time, it should be as high as possible. As already described, the idle speed was set at 60 percent of maximum speed. An additional reduction in consumption is then only possible by rotating the guide vanes ahead of the compressor and the power turbine. Opening the guide vanes ahead of the power turbine causes all the available enthalpy difference to be used to drive the gas generator, and rotating the guide vanes ahead of the compressor has the effect of lowering its power requirement. By means of these two steps, a state of operation with respect to pressure ratio and air and fuel flow was achieved at 60 percent of maximum speed, which, without compressor guide vanes, would not be available until 52 percent of maximum speed.

At the start of acceleration, the vanes can be rotated back to their power position in less than one-tenth of a second. Figure 16 shows how great the influence of compressor guide vane position is on consumption at idle, with power turbine guide vanes (NTL) closed or open.

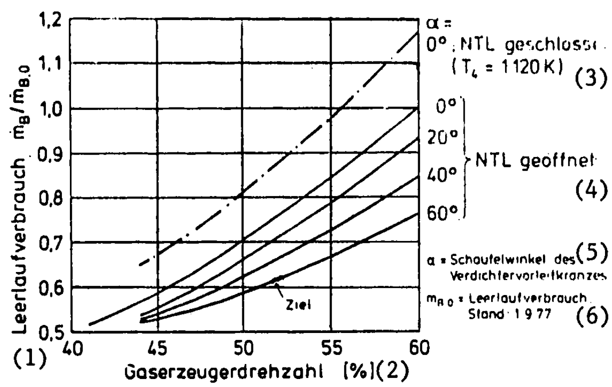

Figure 16. Influence of compressor guide vane position and the opening of power turbine guide vanes on fuel consumption at idle

Key:

1. Consumption at idle, m_B/m_{B0}
2. Gas generator speed [%]
3. NTL closed
4. NTL open
5. Blade angle of the compressor guide ring
6. Consumption at idle, status 1 September 1977

3.3 Emissions

The combustion chambers shown earlier in fig. 11 [2] were tested in the engine. Figure 17 shows the recorded emissions indices for the best diffusion-flame combustion chamber and for the premix combustion chamber. It also shows the limits that must not be exceeded for a given fuel consumption, if particular emissions limits in the U.S. city and highway cycles are to be met. Based on these figures, 1.5 grams/mile NO_x can be achieved with the diffusion-flame combustion chamber. With the premix combustion chamber, 1 gram/mile NO_x can be achieved, using the single-stage type; to achieve 0.4 grams/mile, further development work is needed in the direction of two-stage injection.

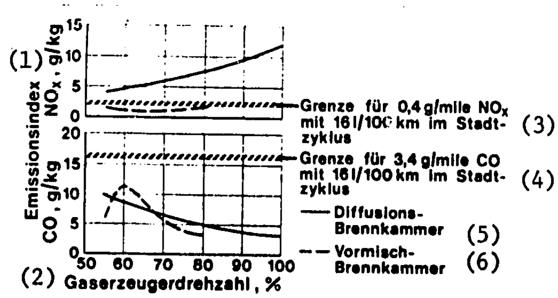

Figure 17. Emissions for diffusion flame and premix combustion chambers

Key:

1. Emissions indices
2. Gas generator speed, %
3. Limit for 0.4 grams/mile NO_x at a consumption of 16 liters/100 kms in the city cycle
4. Limit for 3.4 grams/mile CO with a consumption of 16 liters/100 kms in the city cycle
5. Diffusion-flame combustion chamber
6. Premix combustion chamber

The influence of different fuels on emissions was also tested experimentally. In order to be able to compare the emissions of these different fuels with each other, the energy-specific emissions index was used, which gives the emission of pollutants per energy unit consumed. In fig. 18, the recorded emissions for the different temperature increases in the combustion chamber ΔT_c are shown. The diffusion-flame type with the same injector nozzle was always used as the combustion chamber. These are the results:

1. The NO_x readings show no appreciable differences, except for methanol, where they amount to only 20 percent of the figures for the other fuels. The low adiabatic flame temperature at a stoichiometric ratio and the different fuel-air mixing rate could again be the reason for the difference.

APPROVED FOR RELEASE: 2007/02/09: CIA-RDP82-00850R000500080017-0

2. The CO readings are widely divergent, particularly at low ΔT_c [combustion chamber temperatures]. Methanol and diesel fuel produce a great deal more CO than the other fuels. In the case of methanol, the low temperature and the different fuel-air mixing rate could again be the reason, while in the case of diesel fuel, the unsatisfactory fuel system is responsible for the high CO readings.

3. HC readings demonstrate similar behavior to the CO readings, and the explanation must be be the same as for the CO readings.

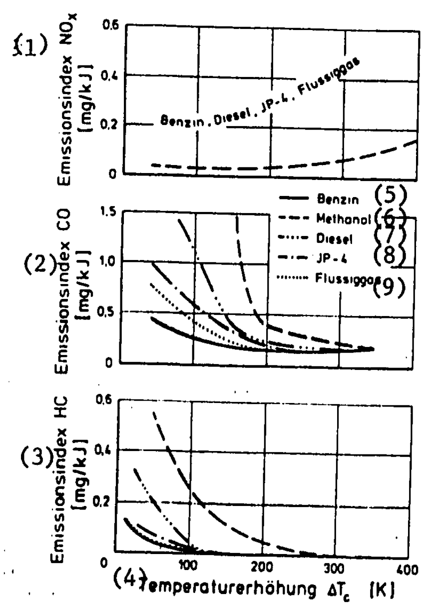

Figure 18. Emissions figures for different fuels

Key:

1. Emissions index NO_x [mg/kJ]	5. Gasoline
2. Emissions index CO [mg/kJ]	6. Methanol
3. Emissions index HC [mg/kJ]	7. Diesel fuel
4. Temperature increase ΔT_c [K]	8. JP-4
	9. Liquid gas

4. Test Vehicle

An Ro 80 was chosen as the test vehicle. It was in production until 1977, manufactured by Audi-NSU, and was normally equipped with a 115 hp Wankel engine. The vehicle, which originally had front-wheel drive, was converted to rear-wheel drive for the installation of the gas turbine and equipped with 3-speed automatic transmission from ZF (Zahnradfabrik Friedrichshafen), without the hydraulic converter. The car has a brake booster and power-assisted steering. With the gas turbine and all the necessary recording equipment on board, the vehicle test weight is 1,700 kg. Air intake is provided by two small lateral grilles, mounted above the fenders. Two tunnels for exhaust gas run under the vehicle's floor to the rear. Heat will be provided by a gas-air heat exchanger, which is supplied with gas taken from the power turbine outlet.

4.1 Behavior on the Road

To demonstrate the good throttle response of the turbine, the vehicle was driven over the U.S city cycle, which is considered representative. Figure 20 shows the traces from the first two phases of this cycle. The speed deviations show how well the driver of the gas turbine test vehicle is able to follow the prescribed speeds and speed changes. As can be seen from fig. 20, the vehicle can adhere to the prescribed speed changes during the greater part of the cycle, with the exception of the quick acceleration at the beginning of the second phase. Future improvements are needed here to achieve a rapid increase in torque, to bring about completely satisfactory road behavior.

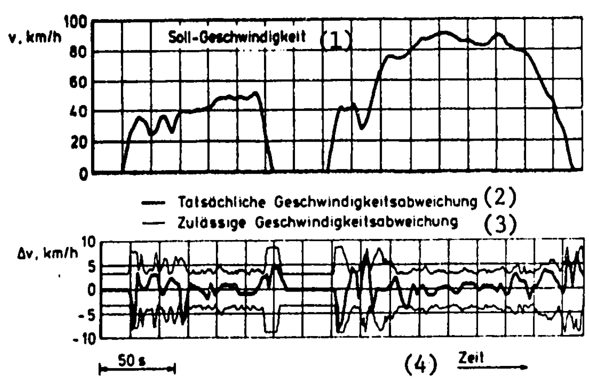

Figure 20. Gas turbine vehicle operation in FDC urban

Key:

1. Prescribed speed
2. Actual deviation in speed
3. Permitted deviation in speed
4. Time

To demonstrate the good power of the engine, which had been measured in test stand experiments, in the vehicle, acceleration times from 0 - 100 km/hour were taken and the distance covered in the first 10 seconds was recorded, accelerating from idle in each case: The results were 13.5 secs and 147 ms.

4.2 Fuel Consumption

The fuel consumption of the experimental gas turbine vehicle was measured in the U.S. city and highway cycles. The results are given in Table 3.

Table 3. Fuel consumption with diesel fuel in U.S. cycles

	City	Highway	Combined
VW-GT 150 in a 1,700 kg Ro 80 test vehicel	15.3 liters/ 100 kms	9.4 liters/ 100 kms	12.6 liters/ 100 kms

In fig. 21, the consumption from the combined cycle is compared with the consumption that was published by the EPA as the U.S. CAFE consumption in 1979. The results show that, in its present state of development, the gas turbine vehicle described here can achieve an overall consumption similar to that obtained by vehicles powered by gasoline engines. Fuel consumption is relatively good at higher speeds; consumption at idle, however, still represents a problem.

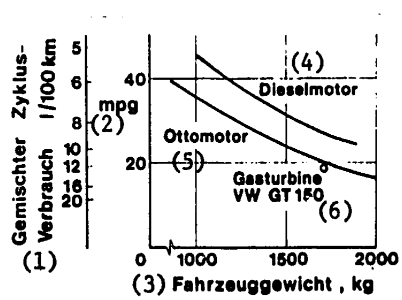

Figure 21. Fuel consumption of the VW-GT 150 and U.S. CAFE in 1979

Key:
1. Combined cycle--consumption in liters/100 km
2. mpg
3. Vehicle weight, kg
4. Diesel engine
5. Gasoline engine
6. VW-GT 150 gas turbine

4.3 Multifuel Capability

Diesel fuel was used in the tests described above. In other tests the engine was operated on gasoline, kerosene or methanol; engine performance remained unchanged. The fuel injection system was adjusted for operation on methanol in accordance with its low heat value.

4.4 Emissions

The optimized diffusion-flame combustion chamber and the single-stage premix combustion chamber were also tested in the experimental vehicle. Table 4 shows the results of emissions readings in the U.S. city cycle. The figure calculated previously from the engine tests were confirmed, namely that the current emissions standards can be met. The two-stage premix combustion chamber, which is necessary to meet the Muskie standards, could not be tested, as it can only be operated with a control system that is not suitable for installation in the vehicle.

Table 4. Emissions in the FDC urban

VW-GT 150 in a 1,700 kg Ro 80 test vehicle	CO grams/mile	HC grams/mile	NO_x grams/mile
With diffusion-flame combustion chamber	3.4	0.3	1.8
With single-stage premix combustion chamber	3.4	0.3	1.0

5. High-Temperature Gas Turbine

Besides the development of components with improved efficiency, increasing the annular process temperature would contribute significantly to an improvement in turbine performance. A temperature increase of 300 K, to 1,650 K, would improve fuel consumption by about 20 percent compared with today's carburetted engines and simultaneously reduce engine volume by about 20 percent. Since this temperature increase would hardly affect reaction zone temperature, the effects on NO_x emissions would also be only slight.

A substantial increase in temperature can only be realized if components made of new high-temperature resistant materials, for example, ceramic, can be successfully developed.

The stresses on materials in a high-temperature gas turbine and on material qualities are outlined in [4].

Figure 23 shows several turbine components manufactured from ceramic material. The materials are silicon nitride and silicon carbide; manufacturing techniques include hot pressing, injection moulding and slip casting. Stationary components, such as the flame pipe and guide vane ring, are the only ones to have been tested so far, at gas temperatures and in temperature cycles like those in a metal gas turbine. Prototype all-ceramic rotors, made from hot-pressed silicon nitride hubs, and reaction-sintered silicon blade rings have withstood circumferential speeds of up to 450 ms/sec in cold-spin tests--measured at the blade tips--and thermo-shock strain up to 500 K/sec.

With rotors made from metal hubs and injection-moulded silicon nitride blades, speeds up to 420 ms/sec have been reached in hot-spin tests. There is still a great deal of scatter in all this development work and unsatisfactory reproducibility of test results. It still appears, however, that, with further applied effort, the development of these components can one day be successfully concluded.

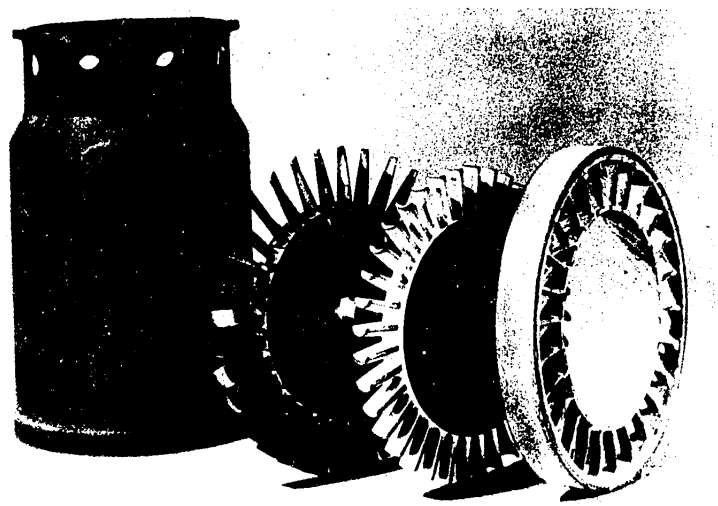

Figure 23. Ceramic prototype components

BIBLIOGRAPHY

1. Walzer, P., Klarhoefer, Ch., and Koehler, M., "Fuel Economy Study of Future Metallic and Ceramic Automotive Gas Turbines," ASME Paper, 76-GT-104.

2. Buchheim, R., "Influences on Exhaust Emissions from Automotive Gas Turbines," ASME Paper , 78-GT-85.

3. Walzer, P., Meier-Grotrian, J., Mardell, J., "An Electronically Controlled Automotive Gas Turbine," ASME Paper, 79-GT-74.

4. Walzer, P., Foerster, S., "Development of Ceramic Components for an Automotive Gas Turbine," JSME, Tokyo, 1977, Paper 67.

5. Walzer, P., Buchheim, R., Meier-Grotrian, J., "Development Work on an Automotive Gas Turbine," 1st Progress Report, Verlag TUeV, Rhineland GmbH.

6. Buchheim, R., Meier-Grotrian, J., "Development Work on an Automotive Gas Turbine," 2nd Progress Report, Verlag TUeV, Rhineland, GmbH.

7. Meier-Grotrain, J., Walzer, P., Rottenkolber, P., "Development Work on an Automotive Gas Turbine," 3rd Progress Report, Verlag TUeV, Rhineland, GmbH.

9581
CSO: 3102/289

END